LECTURE NOTES

ON
DESIGN OF MACHINE ELEMENTS

5TH SEMESTER

PREPARED BY
SIDDHANT SINGH BABU
GUEST FACULTY
DEPARTMENT OF MECHANICAL ENGG

GOVERNMENT POLYTECHNIC, NUAPADA
Government of Odisha
ସରକାରୀ ବହୁବୃତି ଅନୁଷ୍ଠାନ, ନୂଆପଡ଼ା

Machine Design

(1) concept of Machine Design
(2) Riverted Joint
(3) Failure Theories
(4) Bearings
(5) Flat Belt Drive
(6) V-Belt Drive
(7) Gears and Gear Trains

Prepared by →
Siddhant Singh Babu
Techanical Faculty
Adwitiya Academy
Achyra vihar
Bhubaneswar

Machine Design

Basic Concepts of Machine Design

Working Stress

While designing the machine parts it is desirable to keep the stress lower than the maximum or ultimate stress at which failure of the material takes place. The stress is known as working stress. It is also called safe stress.

Factor of Safety

In design it is the Ratio between maximum stress to working stress.

$$F.O.S = \frac{\sigma_{max}}{\sigma_{working}}$$

For a ductile Material it is based upon the yeild point stress

$$F.O.S = \frac{\sigma_Y}{\sigma_w}$$

For a brittle Material which is not properly defined the criteria is ultimate stress. $F.O.S = \frac{\sigma_u}{\sigma_w}$

Failure Theories

As to predict the behaviour of ductile and brittle material in an engineering practice there are certain theories which were investigated in different aspects.

(1) Maximum principal stress Theory (Rankines Theory)

According to this theory Failure occour when the maximum principal tensile stress Reaches the value of the maximum stress at the elastic limit in a simple tension test

$\sigma_1 = \sigma_{et}$ (tension) $\quad \sigma_3 = \sigma_{ec}$ (compression)

It is widely used for brittle material where failure due to brittle fracture takes place.

(2) Maximum shear stress theory (Guest / Tresca Theory)

Failure will occour when the maximum shear stress τ_{max} in a complex system Reaches the value of max shear stress in a simple tension test.

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{\sigma_{et}}{2} \Rightarrow \sigma_1 - \sigma_3 = \sigma_{et}$$

This Theory is widely accepted for ductile Material.

(3) Maximum principal strain theory (St. venant)

Failure of a material occour when the principal tensile strain in the material reaches the strain at the elastic limit in a simple tension test.

$$e_1 = \frac{\sigma_1}{E} - \mu\left(\frac{\sigma_2}{E} + \frac{\sigma_3}{E}\right) \quad \text{so} \quad e_1 > \frac{\sigma_{at}}{E}$$

This theory does not give satisfactory Result about any material. however it gives good result about brittle material for bi-axail tension.

(4) Strain energy Theory (Haigh's Theory)

Failure of a material occour when the total strain energy in the material reaches the total strain energy of the material at the elastic limit in a simple tension test

$$U = \frac{1}{2E}\left[\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\times\mu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)\right] = \frac{\sigma_e^2}{2E}$$

$$\Rightarrow \sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\mu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1) = \sigma_e^2$$

This theory does not gives accurate result but it gives satisfactory Results for ductile Materiale.

(5) Disortion Energy Theory (Mises-henky)

Elastic Failure occour when the shear strain energy per unit volume in the stressed material reaches a value equal to the shear strain energy per unit volume at the elastic limit in a simple tension test.

$$V_s = \frac{1}{12G}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right] = \frac{1}{12G}\sigma_e^2$$

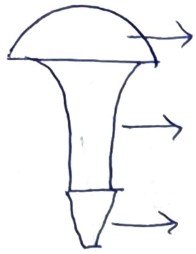
$$= \frac{1}{12G}\left[(\sigma_e - 0)^2 + (0-0) + (0-\sigma_e)^2\right] = \frac{2\sigma_e^2}{12G}$$

$$\text{Now} \quad 2\sigma_e^2 = (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2$$

among all the theories of failure it gives most accurate Result for ductile Materials.

Riveted Joint

A rivet is a short cylindrical bar with head integral to it



The Rivets are used to make Permanet fastening between the Plates, Joints, boilers etc. The main Function of the riveted Joint is to Provide strength and tightness. The Process of forming riveted Joint is called Riveting.

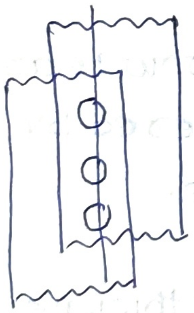
Types of Riveted Joint

(1) Lap Joint → here the two plates that are to be Joined overlap with each other and Joining takes place.

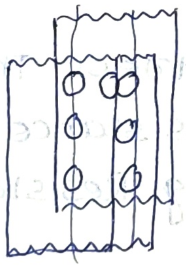
(2) Butt Joint → In this the main parts are kept in alignment with each other and cover plate is Placed on one side or both sides of the Plate. The cover Plate is riveted together.

If only one cover Plate is used is called single strap butt Joint and if it is used on both sides called Double strap butt Joint.

Riveting Procedure



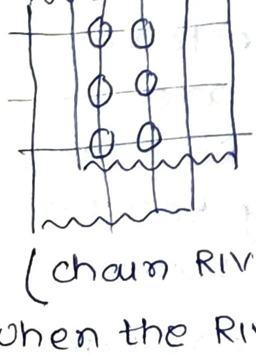
Single riveted Lap Joint



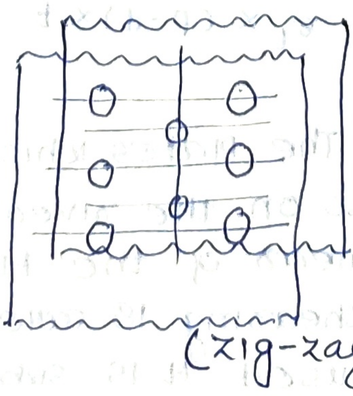
Double riveted Lap Joint

If there is only one row of riveting is called single riveted Lap Joint and so on.

There may be triple riveted or quadruple riveted.



(chain Riveting)



(Zig-zag Riveting)

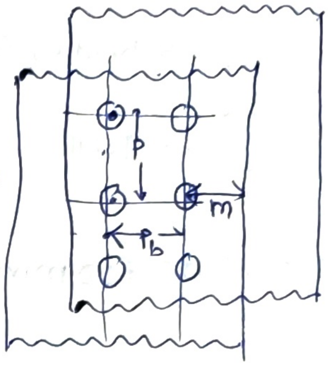
If the rivets in the various rows are staggaed in such a way that every rivet is in the middle of the two rivets of oppsite Rows is called zig-zag Riveting

When the Rivets in various rows are oppsite to each other is called chain Riveting.

Terms used in Riveted Joint

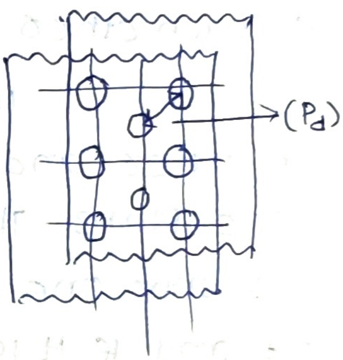
(1) Pitch → distance from centre of one rivet to the centre of the next rivet in the same Row. (P)

(2) margin → distance between centre of rivet hole to the nearest edge of the Plate. (m)



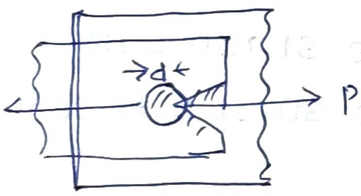
(3) Back Pitch → It is the perpendicular distance (P_b) between the centre line of the sucessive rows

(4) Diagonal Pitch → It is the distance between the centre of the rivets in adjecent rows of zig-zag Riveting. (P_d)



Failure of Riveted Joint

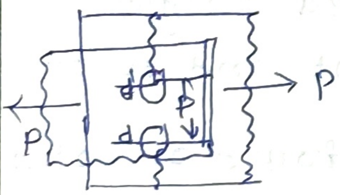
(1) Tearing of the plate at an edge →



This can be advioded by taking margin = 1.5d

d = diameter of Rivet hole

(2) Tearing across the Rows of Rivet →



Due to the tensile stress The row tears along the rivet The Resistance offered by the plate is called tearing Resistance.

here P = Pitch of rivets, d = diameter of rivet hole, t = thickness

σ_t = permisible tensile stress

$A_t = (p-d) \times t$ $\qquad$ $P_t = \sigma_t \times (p-d) \times t$

(3) Shearing of the rivet → The Plates which are connected by rivet exerts tensile stress on the rivets If they are unable to Resists then the shear of the Plane. The Resistance offered by the rivet due to shearing is called shearing Resistance when two cover Plates are used it is subjected to Double shear

τ = permisible shear stress $\qquad$ n = no. of rivets per pitch leng

$A_s = \frac{\pi}{4} \times d^2 \times n$ $\qquad$ $P_s = \tau_s \times \frac{\pi}{4} \times d^2 \times n$

In case of double shear = $2 \times \tau_s \times \frac{\pi}{4} \times d^2 \times n$

(4) Crushing of the Rivets

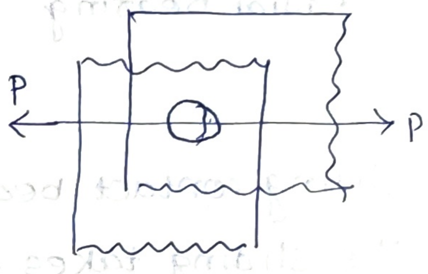
Sometimes the Rivets are not actually shear off the Plane. Due to the crushing of the rivets the rivet hole becomes an oval shaped. The Resistance offered by the plate to the crushing is called crushing Resistance.

d = diameter of rivet hole

t = thickness of the plate

n = no. of rivets per pitch length

σ_c = permisible compressive stress



$A = d \times t \times n$

$P_c = \sigma_c \times d \times t \times n$ (crushing Resistance)

Effciency of the Riveted Joint

The effciency of the riveted joint is defined as the Ratio of strength of riveted joint to the strength of un-riveted solid plate.

$$\eta = \frac{\text{least of } P_t, P_s, P_c}{\text{strength of solid plate}} = \frac{\text{least of } P_t, P_s \text{ or } P_c}{p \times t \times \sigma_t}$$

(*) If in some questions the diameter is not given to us in such cases we use the Unwin's Formula ($d = 6\sqrt{t}$)

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Bearings

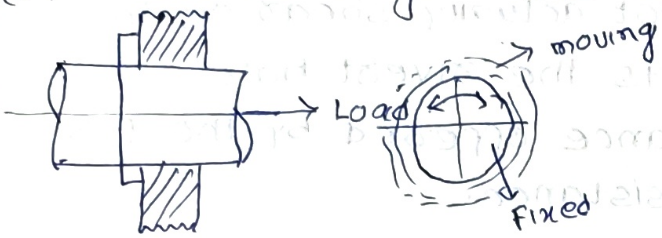
A Bearing is a machine element which support another machine moving element known as Journal. It permists the relative moton between the contact surface of the members while carrying the load. due to this relative motion friction is generated which is to be used by using lubricant. and due to their rapid motion wear also occour. The lubricant create separation between the bearing and Journal.

Classification of Bearing

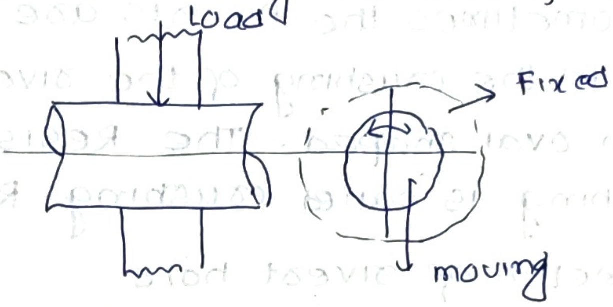
(1) Direction of the load to be supported →

(a) Radial bearings → The load act perpendicular to the direction of motion of the moving element.

(b) Thrust bearings → The load acts along the axis of rotation.



Radial bearing

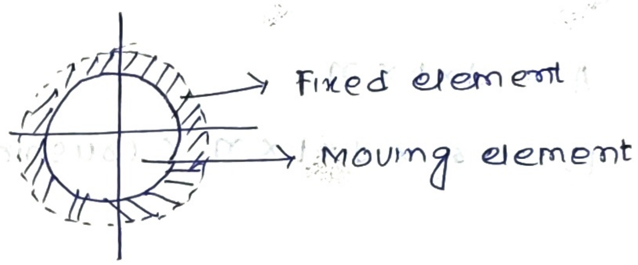


Thrust bearing

(2) Depending upon the nature of contact →

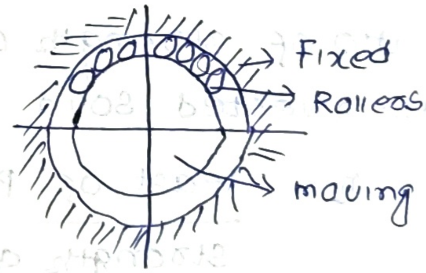
(a) Sliding contact bearing →

The sliding takes place along the surface of contact between the moving and Fixed element. These are also called Plain bearings.



(b) Rolling contact Bearing →

here the steel balls or rollers are interposed between the moving and Fixed element.



Types of sliding contact Bearing

The sliding contact bearing in which the sliding action is along the circumference of a circle or arc of a circle carries radial load are known as Journal or sleeve bearing. when the angle of contact is 360° the bearing is called Full Journal bearing when the angle of contact of the bearing is 120° is called Partial Journal bearing. This type of bearing having less Friction than Full bearing. The Full and Partial bearing may be called as clearance bearing because the diameter of the Journal is less than that of bearing. when an partial Journal bearing has no clearance is called Fitted bearing.

According to this thickness of the layer of lubricant

1) Thick Film Bearing → here the working surfaces are completely separated from each other by lubricant. This type of bearing are called hydrodynamic lubricated bearing.

(2) Thin Film Bearing → The thin Film bearing are those in which, although lubricant is present the working surface are partially in contact with each other.

(3) zero Film Bearing → without any lubricant

(4) hydrostatic / externally pressurized lubricated bearing → here the lubricant is provided externally using high pressure it can support steady load.

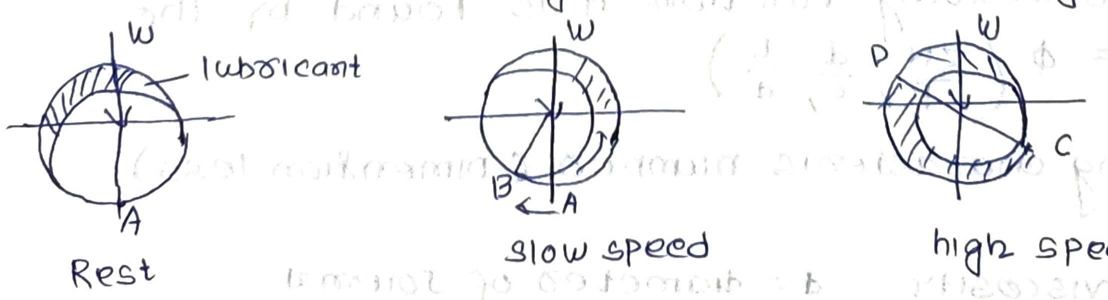
Hydrodynamic Lubricated bearing

As we know there is a thick film of lubricant between the Journal and bearing. when the bearing is supplied with sufficent lubricant a pressure is buid up in the clearance section when the Journal starts rotates in the bearing. The load is supported by the fluid pressure without actual contact between the Journal and bearing. Due to this it can support the load. The Flow of the viscos Fluid is aries due to

1) wedge Film lubrication 2) seqeze Film lubrication

wedge Film Journal bearing

It happens when the Journal/bearing moves relative to the load. The most common case of a steady load. when at Rest position There is a metal to metal contact between the Journal and bearing lubricant is also provided in the clearance space. when the Journal starts rotateing at low speed the Point of contact moved to B But when it rotates at very high speed a continous fluid film is generated. From the point D to C The Film constantly narrowing and hence a converging Film is created called wedge film Journal bearing.

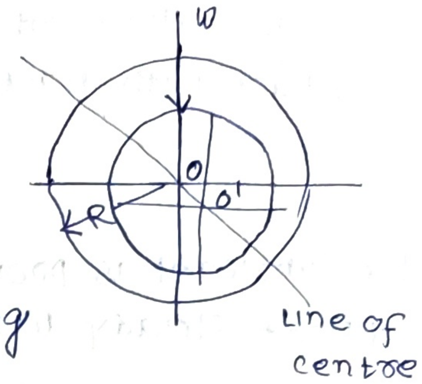


Squeeze Film Bearing

In certain cases the bearing rotates so slowly that the wedge action is not working properly. If the load moves in the Reversed direction then squeeze Film bearing may devlop suffcient capacity to carry dynamic load.

Terms used in Hydrodynamic Journal Bearing

A hydrodynamic Journal Bearing in which O is the centre of the Journal and O' be the centre of the bearing.



D = diameter of bearing   d = diameter of Journal

l = Length of bearing

I) Diametral clearnce → difference between diameter of the bearing and diameter of the Journal.

II) Radial clearnce → difference between radii of bearing and the Journal is called Radial clearnce.

III) Diametral clearnce Ratio → It is the ratio between diametral clearnce to the diameter of the Journal.

IV) Eccentricity → It is the radial distance between the centre (O) of the bearing and the displacement centre (O') of the bearing under load.

V) minimum oil Film thickness → ~~It is the~~ minimum distance between the bearing and Journal under complete lubrication conditions.

VI) Eccentricity Ratio → It is the ratio of eccentricity to the radial clearnce.

VII) Short and long bearing → If the ratio of length to diameter of the Journal is less than 1 is called short bearing otherwise it is called long bearing.

Bearing charasteris Number

The co-effcient of friction in the designing of bearing having a great importance because it affords determing of loss of power due to bearing Friction. It is Found by the experiment $\mu = \phi\left(\frac{ZN}{P}, \frac{d}{c}, \frac{L}{d}\right)$

$\frac{ZN}{P}$ = Bearing charasteris number (Dimention less)

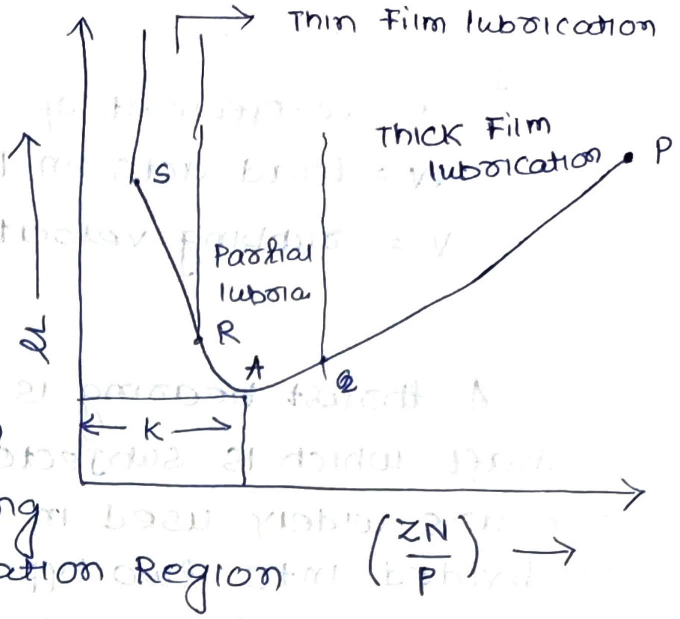
Z = absolute viscosity   d = diameter of Journal

N = speed of Journal   c = Diametral clearnce

P = Bearing pressure = F X A = W X (L X d)

L = Length of Bearing μ = co-effcient of Friction

Experimental investigation



The part of the curve PQ represents the region of thick Film lubrication. Between Q to R the value of μ is very low It is due to the low speed of bearing. So there is partial metal to metal contact between Journal and Bearing. From R to S we get thin Film lubrication Region due to the viscosity of the lubricant ceases to overcome Friction. (PQ → Stable operating, RS → unstable operating condition) From the graph we observe that the minimum amount OF Friction Occours at point A and this point the value of $\frac{ZN}{P}$ is called bearing modulus represent as K. The bearing should not be operated at the point A.

Co-efficient of Friction

The value is $(\mu) = \frac{33}{10^8}\left(\frac{ZN}{P}\right)\left(\frac{d}{c}\right) + K$ K = constant depend upon L/d

Critical pressure of Journal Bearing

The pressure at which the oil film breaks down so that metal to metal contact begins is known as critical pressure or minimum operating pressure.

$$P = \frac{ZN}{4.75\times10^6}\left(\frac{d}{c}\right)^2\left(\frac{L}{d+L}\right)$$

Sommerfeld Number

It is a dimensionless parameter used widely in the design of Journal bearing.

$$S = \frac{ZN}{P}\left(\frac{d}{c}\right)^2$$

c = diametral clearnce
d = diametre of Journal

$$= \frac{ZN}{P}\left(\frac{r}{c_1}\right)^2$$

r = Radius of Journal
c_1 = Radial clearnce.

Heat generated in a Journal Bearing

The heat generated in a bearing is due to the Fluid Friction and Friction of the parts having Relative motion.

$$G_g = \mu \times W \times V$$

μ = co-effecient of friction

W = Load acts on the bearing = $P \times A$

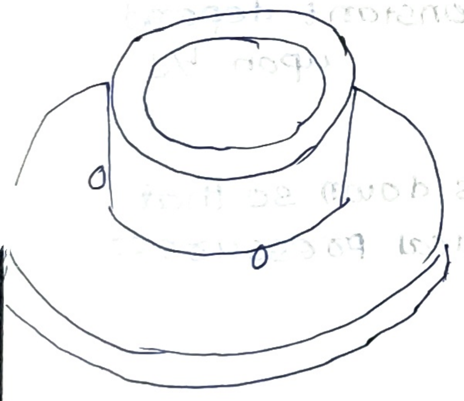
V = rubbing velocity = $\frac{\pi \times d \times N}{60}$

Thrust Bearing

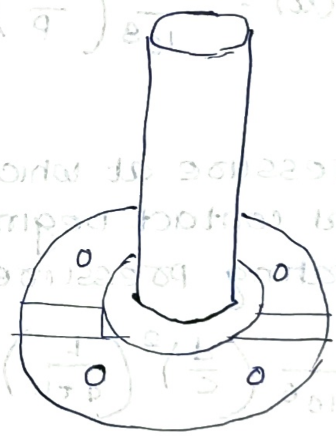
A thrust bearing is generally used to guide or support the shaft which is subjected to a load along it's axis. These are widely used in turbines and propellor shaft. It is divided into two types →

(1) Foot step/ Pivot bearing → The load acts on the shaft is vertical and the end of the shaft rests within the bearing.

(2) collar bearing → The shaft continues throughout the bearing. The shaft may be vertical or horizintal with single collar or multiple collar.



Foot step Bearing



Collar Bearing

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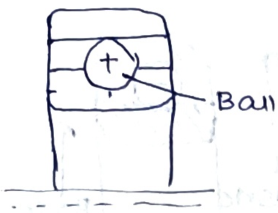
Rolling contact Bearing

here the contact between the bearing surface is rolling instead of sliding. The main advantage of using rolling contact bearing is that it has low starting Friction. Due to the low value of static Friction these are also called antifriction bearing.

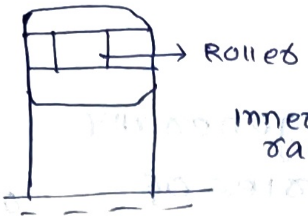
Types of Rolling contact bearings

There are two types of Rolling contact bearing

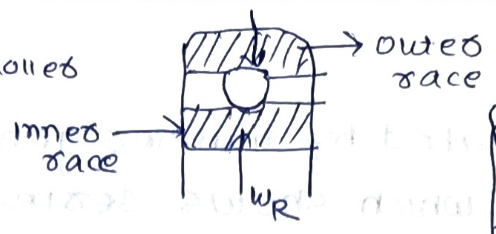
(1) Ball bearing (2) Roller bearing



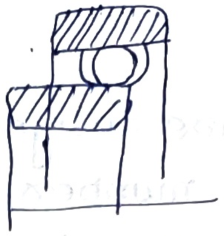
Ball bearing



Roller bearing



Radial load ! Ball bearing



Thrust Ball Bearing

The Ball and Roller bearing consists of inner race which is mounted on the shaft/ journal and outer race is fixed in housing. In between the inner and outer race there are ball or rollers. A no. of balls or Rollers are used and they are placed by some distance by retainers so they do not touch each other. The Ball bearings are used for light loads and Roller bearing are used in heavier load. The Rolling contact bearing depending upon load to be carried are classified as

(a) Radial bearing → supports Radial load and the plane of rotation is normal to the centre line of bearing.

(b) Thrust bearing → It support Radial as well as Thrust load. The action of thrust load is to shift the plane of rotation.

Types of Radial ball Bearings

(i) single Row deep groove bearing → inside the inner and outer races balls are placed by the use of Retainer or cage. These are widely used to high load carrying capacity at high speed.

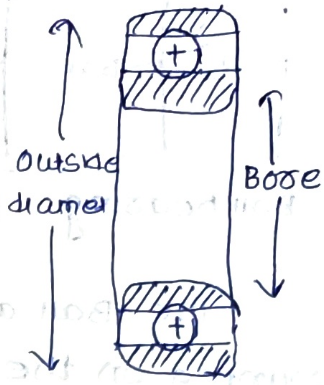
(ii) Filling Notch bearing → These bearing carrying Notches at the inner and outer races so that more no. of balls should be placed than that of deep groove bearing. It has large bearing load capacity.

(iii) Angular contact bearing → These bearings have one side of outer race cut away to permit insertion of more balls than deep groove bearing. which permits to carry a large axail load in one direction while also carries thrust load. These are used in pairs.

(iv) Double Row bearing → These bearings may be either radial or Angular contact which load capacity is slightly less than the single Row bearing.

(v) self aligning bearing → These bearing permit shaft deflection of about 3-5 degree. It is again divided into externally self aligning bearing in which outer race is grounded to spherical surface internal self aligning bearing in which inner outer race is grounded

Designation of Ball Bearing



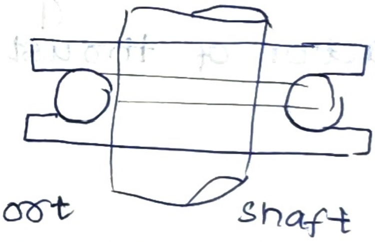
These are designated by number generally a three digit number which shows series of bearing and bore diameter extra number gives the information about it's types
The standard dimension is in milimetres.

1) extra light (100) 2) Light (200) 3) Medium (300) 4) Heavy (400)

For example → 305
That means it is medium series bearing and bore diameter
= 05 x 5 = 25 mm

Thrust Ball Bearing

These are designed to carry thrust load at a speed below 2000 r.p.m. because at higher speed the cetrifugal force tries to force out the balls, in higher speed angular contact bearing are used.



Types of Roller bearing

1) cylindrical Roller bearing → These contains short rollers guided in a cage, these provide lowest co-effcient of friction and used at high speed (Radial motion)

2) spherical roller bearing → These are self aligning bearing balls are in the form of sphere and it can tolerate misalignment of $1\frac{1}{2}^{0}$. It can carry thrust load.

3) Needle Roller bearing → These bearings are relatively slender and completely filled up the space so no retainer is requred. These are used when heavy load are to be used with an oscilatory motion.

4) Tapered Roller bearing → The rollers and race ways are in the form of truncated cones. It can carry radial as well as thrust load. These bearings are used in combinations.

Basic static load Rating

The load carried by a non-rotateing bearing is called satic load and the basic static load rating is the which crossponds to plastc deformation of the balls.

Life of a Bearing

The life of a bearing can be defined as the no. of Revolution or hours at some given constant speed which the bearing runs before the evidence of fatigue failure.

Rating Life

The Rating life of a group of apparentely identical ball or roller bearing is defined as the no. of revolution or hours at some given constant speed that 90% of a group of bearing will survive before the fatigue failure. It is also called minimum life.

Average Life

It is the life of the bearing is defined as the no. of revolution or hours at a constant speed such that 50% of a group of bearing will survive or 50% will fail under operating conditions.

Dynamic Load Rating for Rolling contact bearing

It is given by $L = \left(\frac{C}{P}\right)^K \times 10^6$ Revolution

L = Rating life, P = dynamic load, C = Basic dynamic load Rating = catalogue Rating

$K = 3$ (Ball bearing)

$= \frac{10}{3}$ (Roller bearing)

The Relationship between the life in Revolution (L) and the life in working hours (L_H)

$$L = 60 \times N \times L_H$$

$$C = P \times \left(\frac{L}{10^6}\right)^{\frac{1}{K}} \text{ (Catalgue Rating)}$$

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Flat Belt Drive

The belt drive are used to transmite power from one shaft to another by means of pullers which rotate at same or different speed. The amount of power transmitted of the belt drive depend upon the factors like →

(1) velocity of belt (2) arc of contact of the belt (3) Tension

To obtain maximum power →

(1) The shaft should be properly in line and ensure uniform tension.
(2) The pulley should n't be too close or too apart from each other
(3) A long belt drive can cause swing the pulley from side to side causing the belt to forced out.
(4) The tight side of the belt should be at bottom
(5) The maximum distance should not exceed 10 meters and minimum should n't be less than 3.5 times diameter of the large pulley.

Types of Belt drives

(1) Light drive → velocity Ratio upto 10 m/second. (small machines)
(2) Medium drive → " " From 10 m/s to 22 m/second.
(3) Heavy drive → " " greater than 22 m/second. (generator)

Types of Belt

(1) Flat belt → It is widely used in Factories and workshop where moderate amount of power is being transmitted. From one pulley to another pulley distance is not more 8 meter

(2) V-belt → It is used in Factories and workshop where a great amount of power is being transmitted. But pulley are very near to each other.

(3) circular Belt drive → It is also used in Factories and workshop where great amount of power is being transmitted where the pulleys are more than 8 meters apart.

* Note → when a huge amount of power is being transmitted we can use more than one belt with no. of grooves and obtain very large amount of speed.

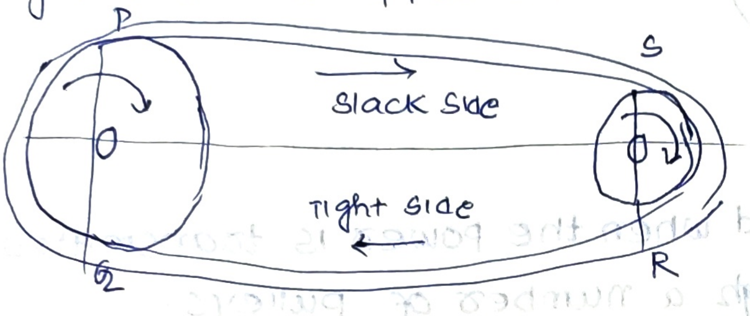
Material used for belt

The material must be strong, flexible and durable. It must have high value of co-efficient of friction

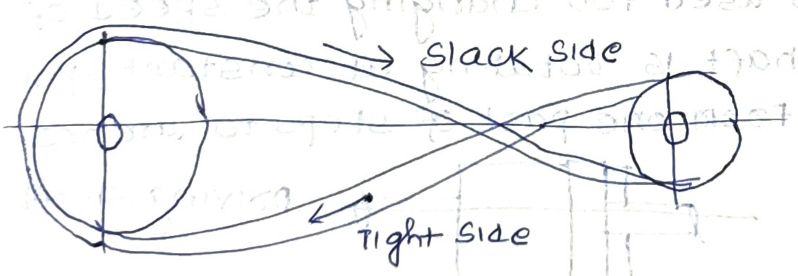
1) Leather Belt 2) cotton / Fabric Belt
III) Rubber Belt 4) Balata Belt

Types of Belt Drives

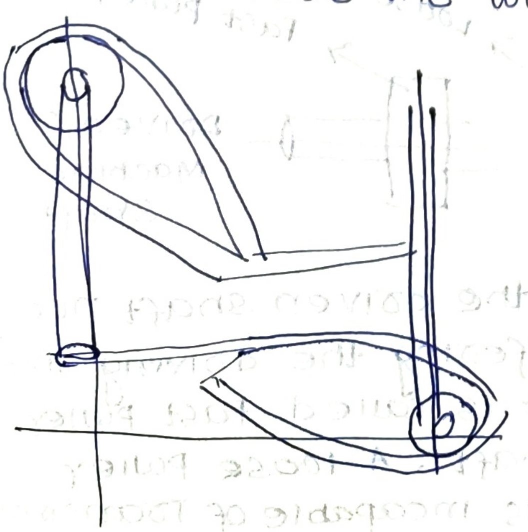
(1) open belt drive → It is used when the shafts are arranged in parallel and rotateing in the same direction. The driver pulley pulls the belt from one side and delivers to the other side Thus the tension in the lower side will be more is called tight side and upper side is called slack side.



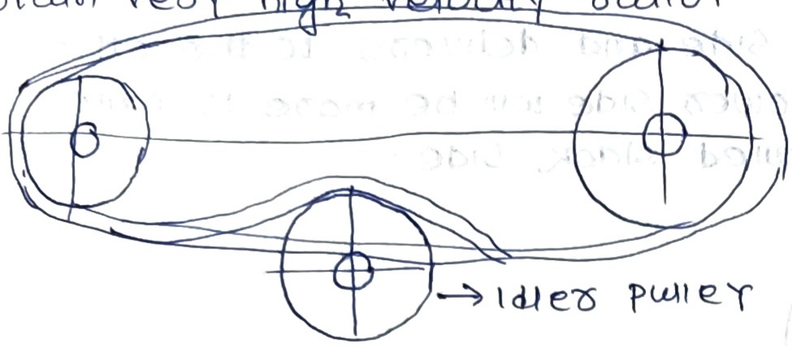
(2) cross Belt / twist belt drive → It is used when the shafts are arranged in parallel and running in oppsite direction. The driver shaft pulls the belt from one side and delivers to the other side. at a point when the belt cross each other it rubs against each other and excessive wear and tear takes place. so speed of the belt is less than 15 m/s and maximum distance of the shaft is 20xb (b = width of pulley)



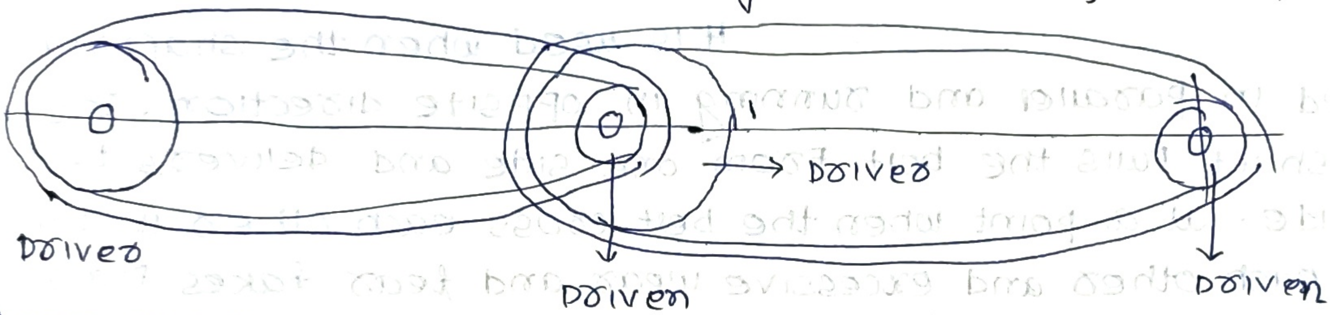
(3) Quarter turn belt drive (Right angle) → It is used when the shafts are arranged at right angle and rotateing in one definite direction. here the width of the face of the pulley = 1.4xb



(4) Belt drive with idler pulley (Jockey pulley) → It is used when when the shaft are in parallel and the open belt drive can't be used to small angle of contact. This type of drive is used to obtain very high velocity ratio.

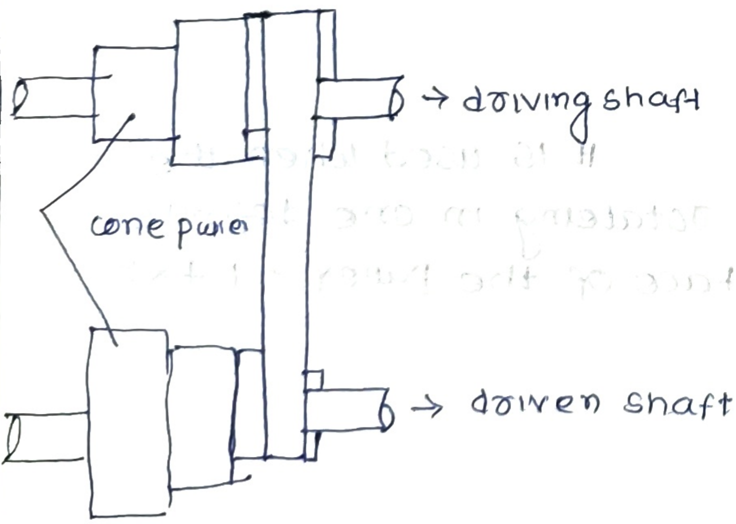


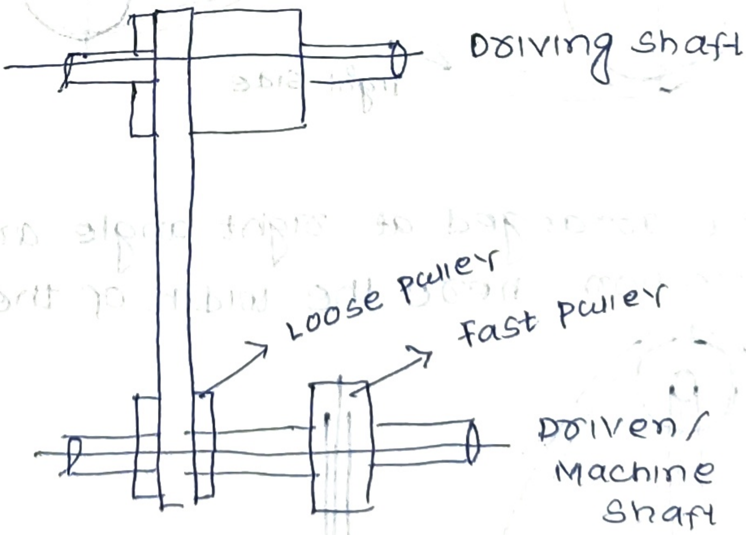
(5) compound belt drive → It is used when the power is transmitted from one shaft to another through a number of pulleys.



here also the shafts are arranged in parallel.

(6) stepped/ cone pulley drive → It is used for changing the speed of driven shaft while the driving shaft is rotating at constant sped. This is done by shifting the belt from one part of steps to another





(7) Fast / loose pulley drive → It is used when the driven shaft needs is to be started or stopped without interfering the driving shaft. A pulley which is keyed to the machine shaft is called Fast pulley and runs at the same speed of driving shaft. A loose pulley runs freely over the machine shaft and is incapable of transmitt any power. when the driven shaft is to be stopped the belt is pused

Velocity Ratio of a Belt Drive

Since the length of the belt passes over both the driver and driven pulley are same.

$$\frac{\pi \times d_1 \times N_1}{60} = \frac{\pi \times d_2 \times N_2}{60} \Rightarrow \frac{N_2}{N_1} = \frac{d_1}{d_2}$$

If thickness of the belt is considered

$$\boxed{\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}}$$

where

N_1 = speed of driving shaft

N_2 = " " driven shaft

d_1 = diameter of the driver pulley

d_2 = " " driven pulley

In case of compound belt drive →

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4} = \frac{\text{speed of last driven}}{\text{speed of first driver}} = \frac{\text{product of drivers}}{\text{product of drivens}}$$

Slip of the Belt

The motion of the belt and pulley is started having a high frictional grip so that the relative motion is started but sometimes the frictional grip becomes insufficient which cause forward motion of the pulley called slip of the belt.

$$\frac{N_2}{N_1} = \frac{d_1}{d_2}\left(1 - \frac{s_1}{100} - \frac{s_2}{100}\right) \Big/ \frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}\left(1 - \frac{s_1}{100} - \frac{s_2}{100}\right) \Big/ \frac{N_2}{N_1} = \frac{d_1}{d_2}\left(1 - \frac{s}{100}\right)$$

s = total slip and always expressed in percentage.

Creep of the belt

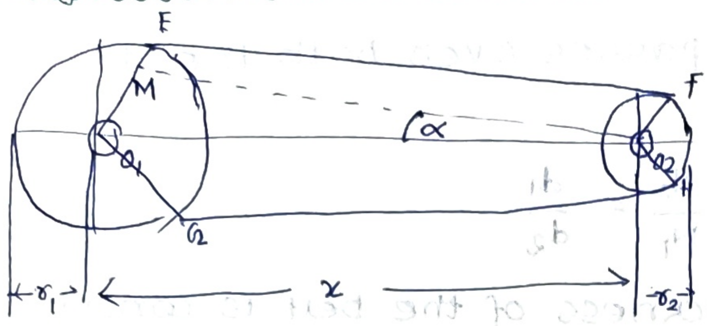
When the belt passes from the slack side to the tight side a certain portion of the belt extends or contract again when the belt moves from tight side to slack side, due to this there is a relative motion between the pulley and belt surface, which is called creep.

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t} \times \frac{E + \sqrt{\sigma_2}}{E + \sqrt{\sigma_1}} \Big/ = \frac{d_1}{d_2} \times \frac{E + \sqrt{\sigma_2}}{E + \sqrt{\sigma_1}}$$

E = Young's modulus of elasticity

σ_1 and σ_2 be the stresses in tight side and slack side

Length of an open belt drive



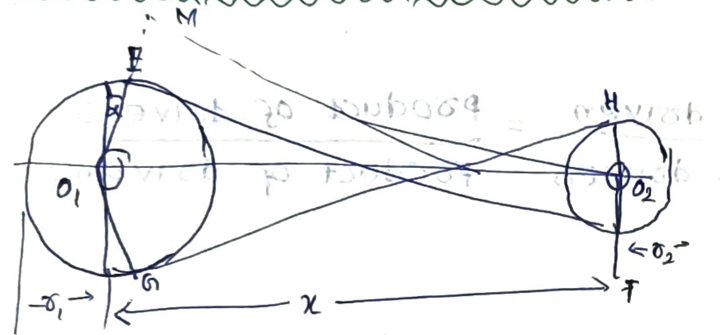
r_1 and r_2 = radii of larger and smaller pulley

x = distance between centre of two pulley

$$\sin\alpha = \frac{O_1M}{O_1O_2} = \frac{r_1 - r_2}{x}$$

$$\text{Length} = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x}$$

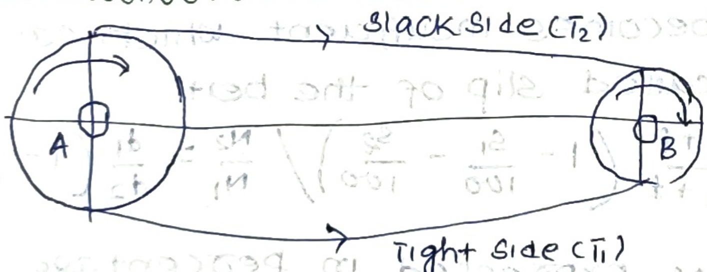
Length of an cross belt drive



$$\sin\alpha = \frac{r_1 + r_2}{x}$$

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$$

Power Transmitted by a Belt



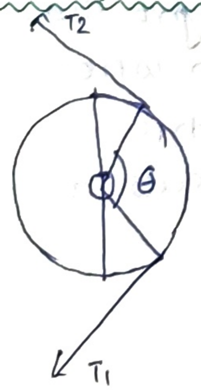
$$P = F \times V$$

$$= (T_1 - T_2) \times V$$

V = Velocity of belt drive

T_1 and T_2 is the tension in tight and slack side

Ratio of driving tension for flat belt drive



T_1 and T_2 is the tension in tight and slack side

μ is the co-efficient of Friction

θ = angle of contact = lap angle

$$\frac{T_1}{T_2} = e^{\mu \times \theta}$$

$\theta = (180^\circ - 2\alpha)$ open belt

$= (180^\circ + 2\alpha)$ cross Belt

Centrifugal Tension

Since the belt is continously runs over the pulleys, therefore some centrifugal force is caused whose effect is to increse the tension in tight and slack side. The tension caused by the centrifugal force is called centrifugal tension.

$T_c = m \times v^2$

m = mass of belt per unit length

v = linear velocity of the belt

When centrifugal force taken into account

$T_{t1} = T_1 + T_c$ and $T_{t2} = T_2 + T_c$

(*) The maximum tension in the belt is always in the tight side of the belt.

Condition for maximum power

$P = (T_1 - T_2) \times v$

as we know $\frac{T_1}{T_2} = e^{\mu \times \theta}$ $\Rightarrow$ $T_2 = \frac{T_1}{e^{\mu \times \theta}}$

$P = \left(T_1 - \frac{T_1}{e^{\mu\theta}}\right) \times v \Rightarrow T_1 \left(1 - \frac{1}{e^{\mu\theta}}\right) \times v \quad \ldots\ldots (1)$

$\Rightarrow T_1 \times v \times C$ where $C = 1 - \frac{1}{e^{\mu \times \theta}}$

As we know $T_{t1} = T_1 + T_c \Rightarrow T_1 = T_{t1} - T_c$

Now putting the value in equation (1)

$P = (T_{t1} - T_c) \times v \times C$

$= (T_{t1} - mv^2) \times v \times C$

$= (T_{t1} \times v - mv^3) \times C$

For maximum power $\frac{dP}{dv} = 0$

$\Rightarrow \frac{d}{dv}(T_{t1} \times v - m \times v^3)$

$\Rightarrow T_{t1} - 3 \times m \times v^2 = 0$

$\Rightarrow T_{t1} = 3mv^2$

$\Rightarrow \boxed{T_{t1} = 3 \times T_c}$

we know $T_{t1} = T_1 + T_c \Rightarrow T_1 = T_{t1} - T_c$

$$= T_{t1} - \frac{T_{t1}}{3} = \frac{2T_{t1}}{3}$$

we can also get the velocity for maximum power

$$\frac{dP}{dv} = 0 \Rightarrow \frac{d}{dv}(T \times v - mv^3) \times C = 0$$

$$\Rightarrow T - 3mv^2 = 0$$

$$\Rightarrow T = 3mv^2$$

$$\Rightarrow \frac{T}{3m} = v^2$$

$$\Rightarrow v = \sqrt{\frac{T}{3m}}$$

—×—×—×

V- Belt drive

We know that v-belt drives are widely used in Factories and workshop where great amount of Power is being transmitt From one pulley to another when the two pulleys are very nearer to each other. The v-belt are generally made up from Fabric and cords with rubber. These belts are mould into trapezodial shape and generally used for short drives. The included angle of the belt is 30° to 40°. The power is being transmitted by the wedgeing action between V-groove in the pulley.

Advantage & Disadvantage of V-Belt over Flat belt

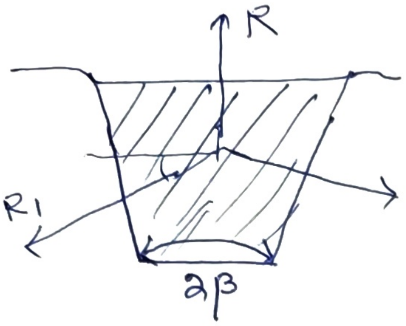
Advantages

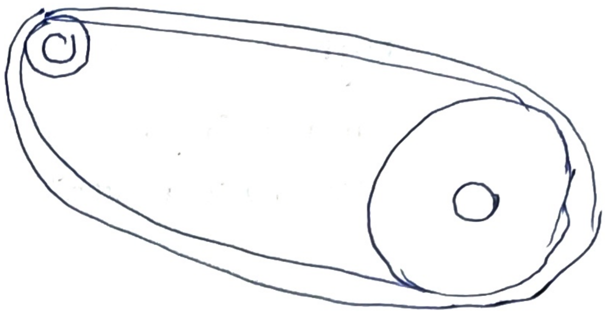
(1) This is a positive drive where slip does not occour.
(2) This gives higher velocity ratio
(3) The value of Friction is less as compared to Flat belt drive
(4) operation of belt and pulley is queit.

Disadvantages

(1) can't be used for large centre distance.
(2) construction of v-belt pulleys are diffcult.
(3) The centrifugal tension is high which prevent it's running.
(4) It also subjected to certain amount of creep.

Ratio of Driving Tension in the belt





V-Belt drive

R_1 = Normal Reaction between the belt and side of the groove

$$R = R_1 + R_2 = 2R_1 \sin\beta \Rightarrow R_1 = \frac{R}{2\sin\beta}$$

$$\text{Frictional Force} = 2\mu \times R_1 = 2\mu \times \frac{R}{2\sin\beta} = \frac{\mu \times R}{\sin\beta} = \mu \times R \times \operatorname{cosec}\beta$$

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu \times \theta \times \operatorname{cosec}\beta$$

$$\Rightarrow \frac{T_1}{T_2} = e^{\mu \times \theta \times \operatorname{cosec}\beta}$$

V-Flat drive

When a Flat belt is replaced by a V-belt it is economical to use Flat-Flaced pulley is called V-Flat drive.

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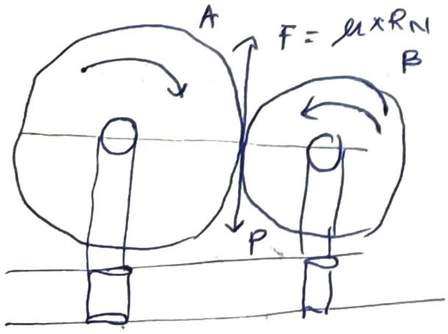
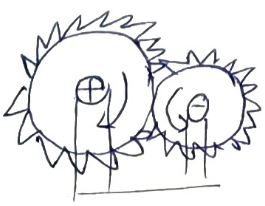
Gear

A gear drive is generally used when the distance between the driver and Follower is very small. The motion and power transmitted by the gear is due to the Frictional wheel or discs.



When the disc mounted on the shaft A rotates the disc B which is keyed to the disc B also rotate but in oppsite direction. In order to avoid slipping between the driver and Follower a number of projections or teeth are cut on the periphery of the wheels. A Frictional wheel with teeth cut on its is known as toothed wheel or gear



(Spur Gear)

Advantage and dis-advantage of Gear drive

Advantages → it transmits exact velocity ratio. higher effciency, transmite large power

disadvantage → manufacturing is diffcult, error in cutting teeth produced vibration.

Classification of Toothed wheel

(1) according to the position of axes of the shaft →

(1) parallel (2) intersecting (3) NON-intersecting & non parallel

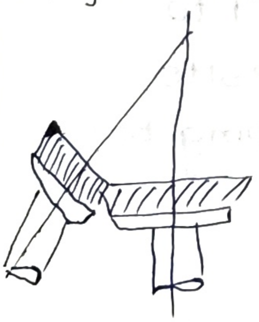
Two parallel and co-planner shaft connected by the gear arrangement is called spur gear arrangement. The teeth are parallel to the axis of the shaft. another name of spur gear is helical gearing in which the teeth are inclined to the axis. The single and double helical gear connecting parallel shaft. The double helical gear is called herringbone gear.

The two non parallel but intersecting shaft where the teeth inclined to the face of the shaft is called bevel gear and when the teeth are inclined to the axes is called helical bevel gear.

The two non parallel and non intersecting shaft and non-coplanner shaft connected by the gear is called spiral gear or skew bevel gear. This type of gear having line contact.

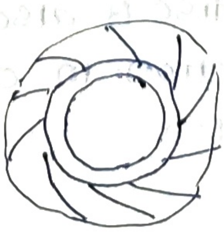
* Note → when equal bevel gears means equal teeth connects two shaft whose axes are perpendicular is called mitres.

A worm gearing is a form of spiral gear in which the shafts are at right angle.

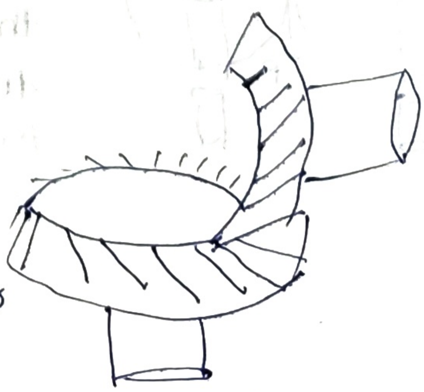


Bevel gear

spiral gear



helical gear



spiral gear

(2) According to the pheripheral velocity →

(a) Low velocity → velocity less than 3 m/second

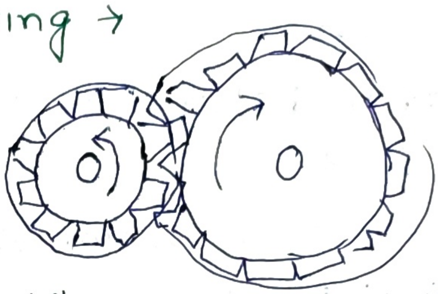
(b) Medium velocity → velocity between 3 to 15 m/second

(c) high velocity → more than 15 m/second.

(3) According to the types of gearing →

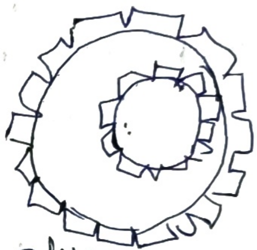
(1) External gearing →

here meshing is done externally larger of two wheel is called spur wheel and smaller wheel is called pinion. The motion of the two wheels are unlike.



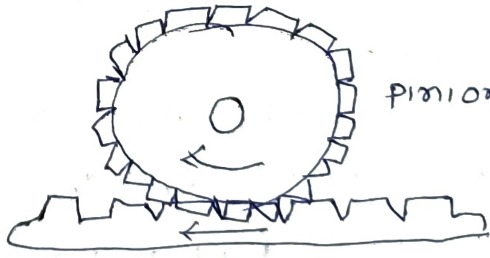
(2) internal gearing →

here meshing is done internally with each other. The larger of the two wheel is called annular wheel and smaller wheel is called pinion. motion of both the wheels are alike.



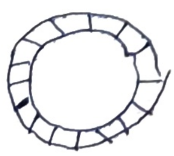
(3) Rack & pinion → sometimes the gear of a shaft meshes external and internally with the gear in a straight line is called Rack and pinion. The straight line gear is called rack and circular wheel is called pinion.

(*) Note → Rack and pinion is used to convert the linear motion to rotary motion or vice-versa.



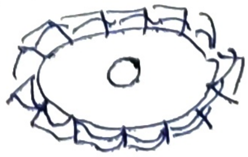
(4) According to the teeth on the gear surface →

(1) straight teeth



(2) inclined teeth

(3) curved teeth



Terms used in Gears

Pitch circle → It is an imaginary circle which by pure rolling motion, would give the same motion of actual gear. The diameter of the pitch circle is called pitch circle diameter and the size of the gears are usually specified by pitch circle diameter.

Pitch point → common point of contact between two pitch circles.

Pressure angle / angle of obliquity → It is the angle between the common normal to two gear tooth and common tangent at pitch point. The standrad pressure angle is $14\frac{1}{2}°$ and $20°$.

Addendum → Radial distance of a tooth from the pitch circle to the top of the tooth.

Dedendum → Radial distance of a tooth from pitch circle to the bottom of the tooth.

Addendum circle → circle drawn through the top of the teeth and concentric with pitch circle.

Dedendum circle → circle drawn through the bottom of the teeth

Circular pitch → distance measured on the circumference of the pitch circle from a point of one tooth to the crossponding point of another tooth. $P_c = \frac{\pi D}{T}$ D = diameter of pitch circle

T = Teeth on the wheel

Diametral pitch → It is the ratio of the number of teeth to the pitch circle diameter in milimeters. $P_d = \frac{T}{D}$

so $P_c \times P_d = \pi$

Module → It is the ratio between pith circle diameter in milimeter to the number of teeth. $(m) = D/T$

Clearance → Radial distance from the top of the tooth to the bottom of the tooth.

Total depth → Radial distance between the Addendum and dedendum circle of a gear ($A_m + d_m$)

Working depth → Radial distance from the Addendum circle to the clearance circle.

Path of contact → path traced by the point of contact of the two teeth from the begining to end of engagement.

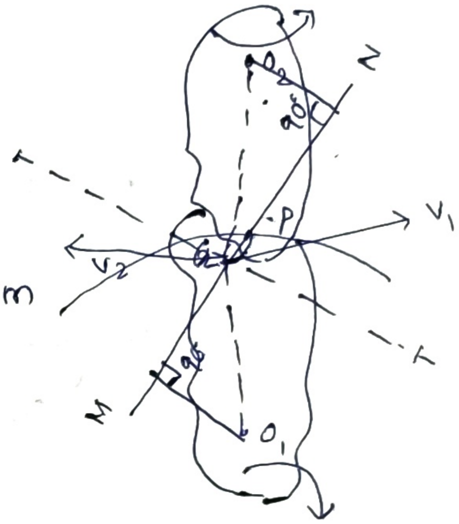
Length of arc of contact → It is the length of the common normal cut off by the Adendum circle of the wheel and pinion.

Arc of contact → Path traced by the point on the circle from the begining to the end is called arc of contact. beginging to the pitch point is called approach and from the pitch point to the end is called path of recess.

Law of gearing

The common normal drawn at the point of contact between a pair of teeth must always pass through the pitch point. In order to get a constant velocity ratio all gear system must satisfies law of gearing.

$$\frac{w_1}{w_2} = \frac{O_2P}{O_1P}$$

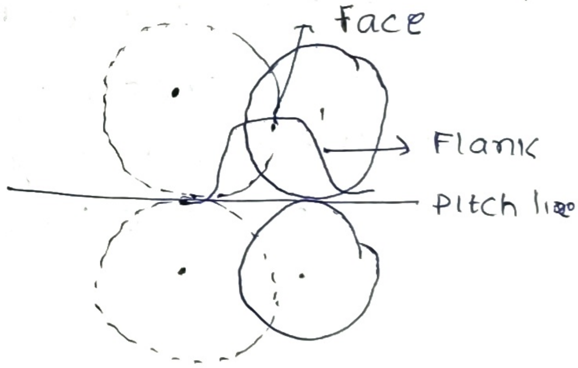


The angular velocity ratio is inversly proportional to the Ratio of distance from Pitch point.

Forms of Teeth

The two types of teeth are commonly used in the gearing arrangement:

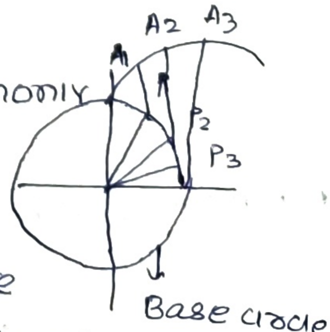
(1) Cycloidal Teeth →



A cycloid is the curve traced by a point on the circumference of a circle which rolls without slipping along a straight line.

(2) Involute Teeth → An involute of a circle is a plane curve generated by a point on a tangent which rolls on the circle without slipping in connection with the circle is called involute gear teeth.

Comparision



In actual practice involute gear are more commonly used as compared to cyclodial gears because in case of involute gear the centre distance betn a pair of gear tooth can be varied without varying the velocity ratio. The pressure angle is Remains constant throughout the engagement which gives less wear. Involute teeth are easy to manufacture because it requires single curve for designing of face and flank whereas in the cyclodial teeth it requires two circle for manufacturing.

System of gear teeth

(1) $14\frac{1}{2}^0$ composite system → general purpose gears. The tooth Profile of this system are cycloidal curve at the top and bottom and involute at the middle portion.

(2) $14\frac{1}{2}^0$ Full depth involute system → It is used in spur and helical gear

(3) 20^0 Full depth involute system → Due to this increse in Pressure angle the tooth is very strong

(4) 20^0 stub involute system → very strong and used in heavy load.

(*) contact Ratio → $\frac{\text{Length of arc of contact}}{\text{circular pitch}}$

interference in involute Gears

The phenomenon when the tip of tooth undercut the root on it mating gear is known as interference.

interference may only be prevented if the addendum circle of the two mating gear cut the common tangent to the base circle betn the point of tangency.

Minimum number of teeth on the pinion to avoid interference

$$t = \frac{2 \times A_p}{\sqrt{1 + \frac{T}{t}\left(\frac{T}{t}+2\right)\sin^2\phi} - 1}$$

so the teeth for different Pressure angle

$14\frac{1}{2}^0$ composite = 12

$14\frac{1}{2}^0$ Full depth involute = 32

20^0 Full depth involute = 18

20^0 Stub involute = 14

minimum teeth on the wheel

$$T = \frac{2 \times A_w}{\sqrt{1 + \frac{t}{T}\left(\frac{t}{T}+2\right)\sin^2\phi} - 1}$$

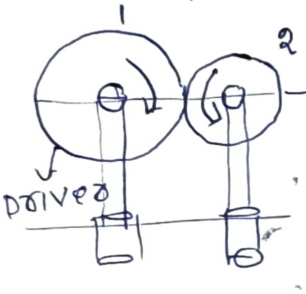
×××

Gear Trains

Sometimes two or more gears made to mesh with each other to transmite power from one shaft to another is called gear trains.

Types of gear train

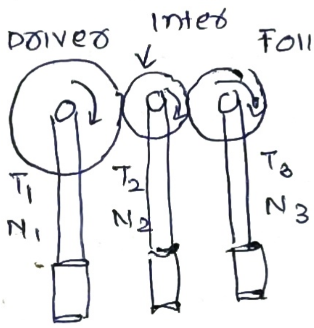
(1) Simple gear train → when there is only one gear mounted on each shaft is known as simple gear train. These gears are represent by their pitch circle.



speed ratio $\frac{N_1}{N_2} = \frac{T_2}{T_1}$

T_1 and T_2 be the teeth in driver and driven shaft. and N_1 and N_2 be the speed of driver and driven shaft.

Train value = $\frac{N_2}{N_1} = \frac{T_1}{T_2}$

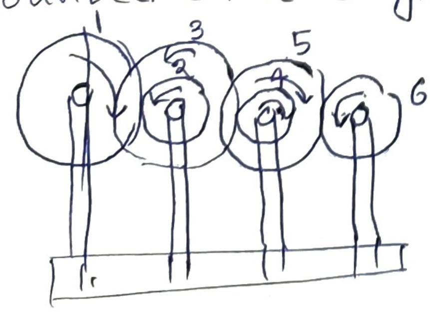


$\frac{N_1}{N_2} = \frac{T_2}{T_1}$, $\frac{N_2}{N_3} = \frac{T_3}{T_2}$

$\Rightarrow \frac{N_1}{N_3} = \frac{T_2}{T_1} \times \frac{T_3}{T_2} = \frac{T_3}{T_1}$

If the number of intermediate gears are odd then the motion of driver and driven is alike and if the gears are even their motion is oppsite.

(2) compound gear train → when there are more than one gear mounted on a single shaft it is called compound gear train.

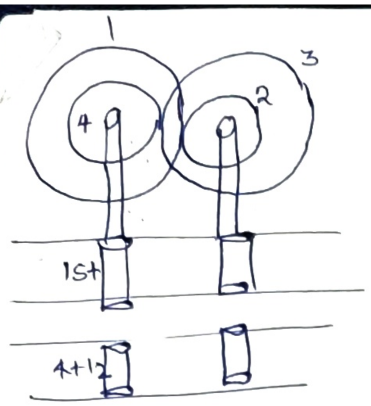


$$\frac{\text{speed of 1st driver}}{\text{speed of last driven}} = \frac{\text{no. of Teeth on drivens}}{\text{no. of teeth on driver}}$$

$$\frac{N_1}{N_6} = \frac{T_2 \times T_4 \times T_6}{T_1 \times T_3 \times T_5}$$

when gears are mounted on the same shaft their speed is equal so $N_2 = N_3$ and $T_4 = T_5$ and their direction of motion is also same.

(3) Reverted Gear Train → when the axes of the first gear and last gear are co-axail then the gear train is called as reverted gear train.



here

r_1, r_2, r_3 and r_4 is the pitch circle radius of gear 1, 2, 3, and 4.

$$r_1 + r_2 = r_3 + r_4$$

Similarly $T_1 + T_2 = T_3 + T_4$

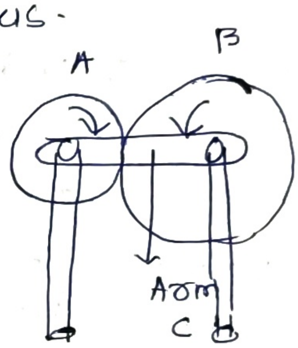
because the centre distance between gear 1 and 2 is equal to centre distance of 3 and 4

$$\frac{N_1}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}$$

Epicyclic Gear Train

The axes of the shaft over which the gears are mounted may move relative to a fixed axis.

When any of the gear A, B and arm is fixed the other two rotates about that axis and that motion is called epicyclic and the gear train is called as epicyclic gear train.



Let us consider the arm is fixed in this diagram so here the speed of arm = 0 ($N_C = 0$)

Speed of gear A w.r.to arm C = $N_A - N_C$

Speed of gear B w.r.to arm C = $N_B - N_C$

$$\frac{N_B - N_C}{N_A - N_C} = -\frac{T_A}{T_B}$$

(oppsite direction is due to external meshing)

examples of Gear Trains

gear box, lathe machines, press machine → Simple gear train

watches, clocks, ships → compound gear train

lathe back gears, clocks → Reverted gear train

Automotive transmission → epicyclic gear train

—x—x—x—

Questions of Machine Design

(1) The Ratio of ultimate stress to the design stress is known as →

(a) Elastic limit (b) Plasticity (c) F.O.S (d) strain

(2) In a body a thermal stress is one which arises due to →

(a) Latent heat (b) specific heat (c) heat (d) Temperature gradient

(3) Rankine theory is used for →

(a) Ductile Material (b) Brittle (c) semi-brittle (d) Elastic material

(4) Guest theory is used for →

(a) Ductile (b) Brittle (c) Elastic (d) Plastic Material

(5) according to unwins Formula The Relation betn diameter of the rivert hole (d) and thickness of the Plate (t) is given by

(a) $d = 6\sqrt{t}$ (b) $d = 2t$ (c) $d = 1.6t$ (d) $d = 2.1t$

(6) A lap Joint is always in ____ shear.

(a) single (b) double (c) No shear (d) tension

(7) If the tearing effciency of a riverted Joint is 50% then what is the Ratio of d/p is

(a) 0.33 (b) 0.50 (c) 0.90 (d) 1.00

(8) according to the indian standrad the diameter of the rivert hole for a 24 mm diameter rivert is

(a) 20 mm (b) 23 mm (c) 25 mm (d) 26 mm

(9) In case of centrifugal tension in a Flat belt drive what is the condition for maximum power?

(a) $T = T_c$ (b) $T = 2T_c$ (c) $T = 3T_c$ (d) $T = 4T_c$

(10) All the streses produced in a Flat belt drive are

(a) compressive (b) Tension (c) Both (d) shear

(11) For maximum power the velocity of the belt is given by

(a) $\sqrt{\frac{T}{m}}$ (b) $\sqrt{\frac{T}{2m}}$ (c) $\sqrt{\frac{T}{3m}}$ (d) $\sqrt{\frac{T}{4m}}$

(12) The included angle of V-belt is

(a) 20°-30° (b) 30°-40° (c) 40°-50° (d) 50°-60°

(13) The groove angle of the pulley of V-belt drive is

(a) 20° - 25° (b) 25° - 32° (c) 32° - 38° (d) 38° - 45°

(14) In a Full Journal bearing the angle of contact between the bearing and Journal is

(a) 90° (b) 180° (c) 270° (d) 360°

(15) In case of thrust bearing the load acts

(a) along the axis of rotation (b) parallel to the axis of rotation

(c) perpendicular to the axis of rotation

(16) The rolling contact bearings are called as ______ Bearing.

(a) thin lubricated (b) thick lubricated (c) anti-Friction (d) NONE

(17) The size of the gears are usually specified by →

(a) pressure angle (b) pitch circle diameter (c) circular pitch

(18) The minimum no. of teeth on the pinion in order to avoid interference for a 20° stub system is ______

(a) 12 (b) 14 (c) 18 (d) 24

(19) The gears are termed as medium velocity gear if their peripheral velocity is

(a) 1-3 m/s (b) 3-15 m/s (c) 15-30 m/s (d) 30-50 m/s

(20) In a simple gear train the driver and Follower having 16 and 20 Teeth. If the speed of driving shaft is 200 r.p.m then what is the speed of driven shaft?

Ans → As we know in case of simple gear train

$$\frac{N_1}{N_2} = \frac{T_2}{T_1}$$

$$\Rightarrow N_2 = \frac{T_1}{T_2} \times N_1 = \frac{16}{20} \times 200 = 160 \text{ r.p.m}$$

(21) In a compound gear train if a gear B is mounted on shaft contains another gear C and if B rotates in the clockwise direction then what is the rotation of gear C?

(a) clockwise (b) anti-clockwise (c) Both (d) Fixed